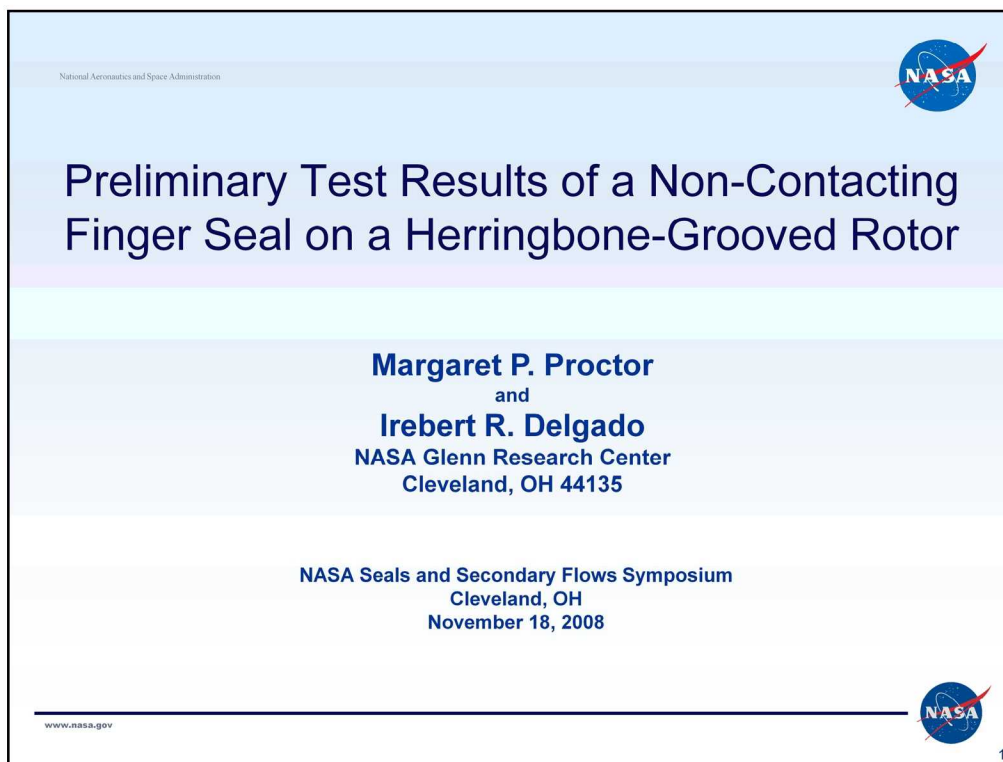# PRELIMINARY TEST RESULTS OF A NON-CONTACTING FINGER SEAL ON A HERRINGBONE-GROOVED ROTOR

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This is nearly the same presentation I made in July 2008 at the Joint Propulsion Conference in Hartford, CT. The details of this work can be found in the NASA TM-2008-215475 and AIAA-2008-4506.

Low leakage, non-contacting finger seals have potential to reduce gas turbine engine specific fuel consumption by 2 to 3 percent and to reduce direct operating costs by increasing the time between engine overhauls.

To investigate the potential of the non-contacting finger seal and to provide data to develop a verified design methodology for it, a baseline non-contacting finger seal was designed and fabricated. Static tests and initial spin tests have been conducted. The test hardware, apparatus, procedures as well as the leakage performance, power loss, and wear results will be presented. Unexpected findings prompted exploratory bind-up tests not typical of previous test procedures.

Baseline Non-Contacting Finger Seal
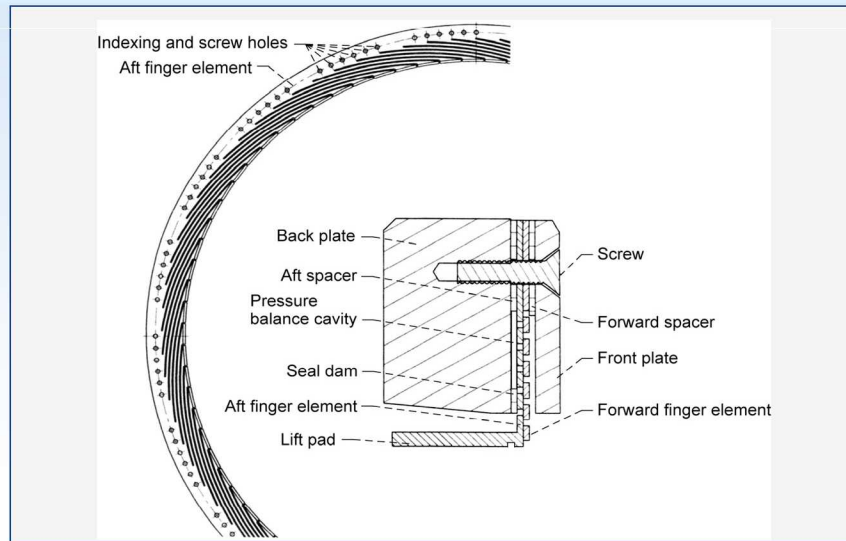
The baseline non-contacting finger seal is a NASA patented design. The primary difference between it and Gul Arora's design patented by AlliedSignal is that there are no lift pads on the high pressure fingers.

The baseline non-contacting finger seal is comprised of a back plate, aft spacer, aft (or low pressure) finger element, forward (or high pressure) finger element, forward spacer, and front plate. The components are held together with 20 flat head screws. A typical seal would have a back plate of approximately the same thickness as the front plate and would be riveted together. The thicker back plate allows use of threaded fasteners so that different finger elements can be tested without having to replace all the individual seal components. The finger elements are essentially washers made of thin sheet stock with multiple curved slots machined around the inner diameter to form the fingers. They are clocked so that the fingers of one cover the slots of the other. The aft finger element fingers have axial extensions or "lift pads" at the seal id that are concentric to the rotor. The fingers act as cantilever beams and flex in response to rotor dynamic motion and radial growth of the rotor due to centrifugal or thermal forces.

Non-Contacting Finger Seal—Pre-Test

Circumferential groove
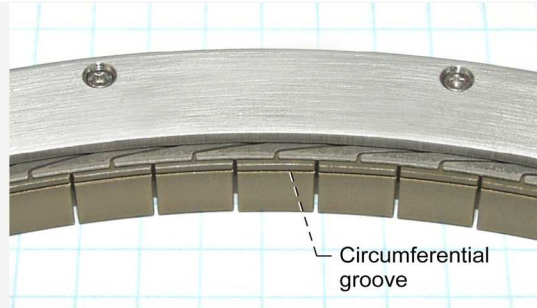
- Haynes–188
- Temperatures up to 1089 K
- Radial clearance to rotor = 25.4 µm (0.001 in)
- Lift pads ride over herringbone grooves

Another difference between the forward and aft fingers elements is that the high pressure fingers have a larger inner diameter to ensure they don't touch the rotor due to pressure blow down effects. Applying a pressure differential across a finger seal generates a suction force that draws the fingers towards the rotor due to the lower pressure under the finger pads. It's possible to reduce the high pressure finger element id to match the low pressure finger element id if there is sufficient friction between the two elements to keep them moving together.

The lift pads have a circumferential groove so that low pressure exists at all four edges of the lift pad.

Herringbone Grooves on Seal Test Rotor—Pre-Test

Rotation
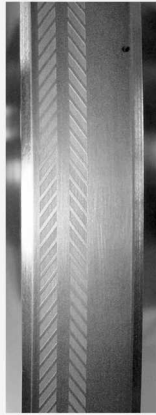
- Rotor O.D.: 21.6 cm (8.5 in)
- Grainex Mar-M–247 rotor
- Chrome carbide coating (HVOF)
- Surface finish: 0.2 µm (8 µin)
- 536 grooves (268 around circumference)
- Groove depth: 20 µm (0.0008 in)
- Groove ends:
  – Begin at middle of circumferential groove on lift pads
  – Extend past low pressure edge of lift pads

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The direction of rotation causes pumping towards the center of the groove pattern.

High-Temperature, High-Speed Turbine Seal Rig

The High-Temperature, High-Speed Turbine Seal Rig is used to measure seal leakage and power loss at different speeds, pressure differentials and temperatures.

A torque meter is used to measure the seal torque. Tare torque is measured without a seal and is subtracted from the torque measured with a seal installed to determine the seal torque. Seal power loss is simply the seal torque multiplied by the speed. When a pressure differential is applied across the seal windage on the high pressure side of the test disk and balance piston increases due to the increased density of the air. Bearing torque is also increased. The additional windage and bearing torque are approximated and subtracted from the measured power loss. Hence the seal power loss presented is approximate.

Test Seal Configuration and Location of Research Measurements
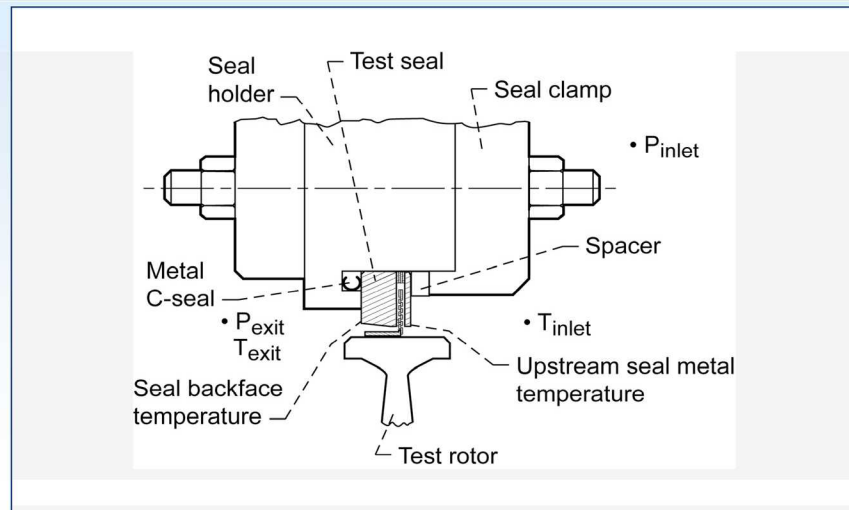
The seal inlet and exit pressures and temperatures and the seal backface temperature are measured at three equally spaced locations around the circumference.

Flow Factor

$$\phi = \frac{\dot{m}\sqrt{T_{avg}}}{P_u \times D_{seal}} \quad , \quad \frac{kg\text{-}\sqrt{K}}{MPa\text{-}m\text{-}s}$$

$\dot{m}$ = air leakage flow rate, kg/s.

T_{avg} = average seal air inlet temperature, K.

P_u = air pressure upstream of seal, MPa.

D_{seal} = outside diameter of the test rotor, m.

The seal leakage rate is used to compute the flow factor.

Test Procedure

| Test	Shaft speed, rpm	Inlet air temperature, K	Pressure drop across seal, kPa	Comment
Initial static tests	0	300 533 700	0–517 kPa (or max.) – 0	
Shaft bound. Seal removed, visually inspected and reinstalled.				
Bind-up tests Part 1 Part 2	0	300 K	69, 138, 207, 276, 345, 414, 483 0-276 in 13.8 increments, 483, 552, 576	Can shaft turn by hand? at 0 kPa at test pressure
Seal removed for visual inspection. Deposits on seal sampled. Seal ultrasonically cleaned and reinstalled.				
Repeat Static Tests	0	300 533 700	0 – max. – 0, 3 cycles	
Shaft bound. Seal removed, visually inspected, and reinstalled.				
Spin Test 1	0 5,000 0 0	300 K ↓	13.8 13.8, 34, 69, 103, 138, 103, 69, 34, 13.8 13.8 0	25 min at 5000 rpm
Seal removed, visually inspected, and reinstalled.				
Spin Test 2	0 0 5,000 0	300 K ↓	0 – max. – 0, 1 cycle 13.8 13.8, 34, 69, 103, 138, 172, 206, 241, 206, 172, 138, 103, 69, 34, 13.8 13.8	68 min at 5000 rpm
Seal removed and visually inspected.				

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Self explanatory.

The seal bound up the shaft at pressure differentials of 276 KPa (40 psid) and higher.

The shaft was found bound by seal after the repeat static test and rig was cooled down. This can be explained. The pressure differential used during cool down was 345 KPa, which is higher than the pressure at which the seal binds the shaft. Since the clearance increases with temperature and the pressure locked the seal onto the shaft, when the rig cooled down it trapped the seal in the locked position.
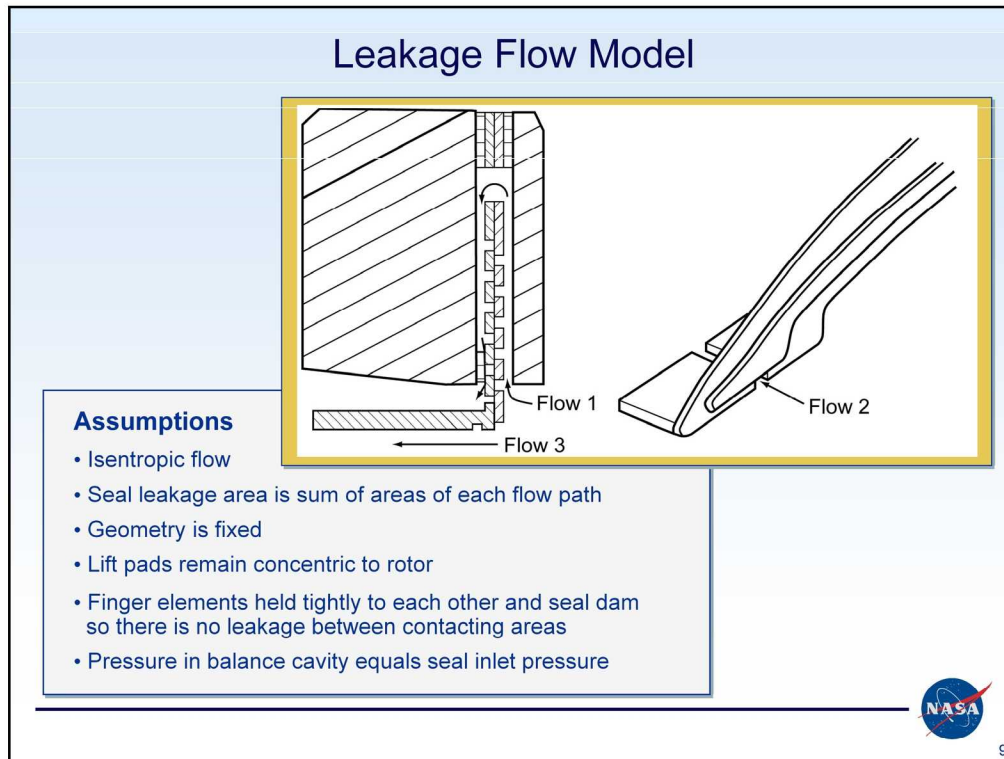
Leakage Flow Model

A simple leakage model was used to predict the seal leakage rate.

The model assumes there are three flow paths through the seal as shown.

The sum of these three areas equals the seal leakage area.

The assumption that the pressure in the balance cavity equals the seal inlet pressure is good because the flow area of the finger slots at the seal dam is 11 times smaller than the area of the flow restrictions upstream of it. This means the finger slots control the leakage rate in that flow path.

Leakage Flow Model

$$\dot{m} = \frac{P_u}{\sqrt{RT_u}} \cdot A\sqrt{\gamma}M\left(1+\left(\frac{\gamma-1}{2}\right)M^2\right)^{1/2-\frac{\gamma}{\gamma-1}}$$

where

$$M = \left[\left(\left(\frac{P_u}{P}\right)^{\frac{\gamma-1}{\gamma}}-1\right)\frac{2}{\gamma-1}\right]^{\frac{1}{2}}$$

For air (γ = 1.4), when $P/P_u \leq 0.5283$ the flow is choked

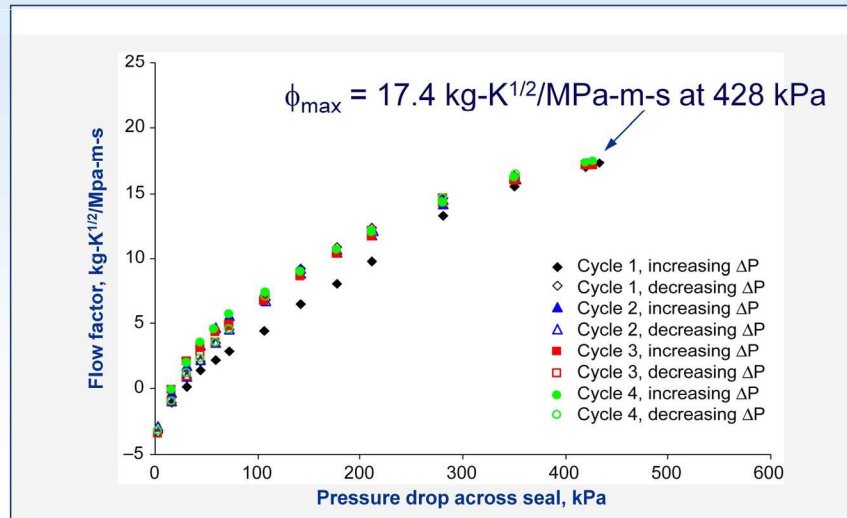$$\dot{m} = \frac{P_u}{\sqrt{RT_u}} \cdot A \cdot (0.6847)$$

The leakage rate is predicted using the isentropic flow equation.

A discharge coefficient of 0.65, which is typical of orifices, can be applied to account for inlet and exit losses.

Initial Static Leakage Performance at 300 K
Radial Clearance = 25.4 µm

ϕ_{max} = 17.4 kg-K$^{1/2}$/MPa-m-s at 428 kPa

The maximum flow factor of approximately 17.4 kg-K$^{1/2}$/MPa-m-s occurred at approximately 428 kPa across the seal. The data show little hysteresis after the first cycle of increasing pressure differential.

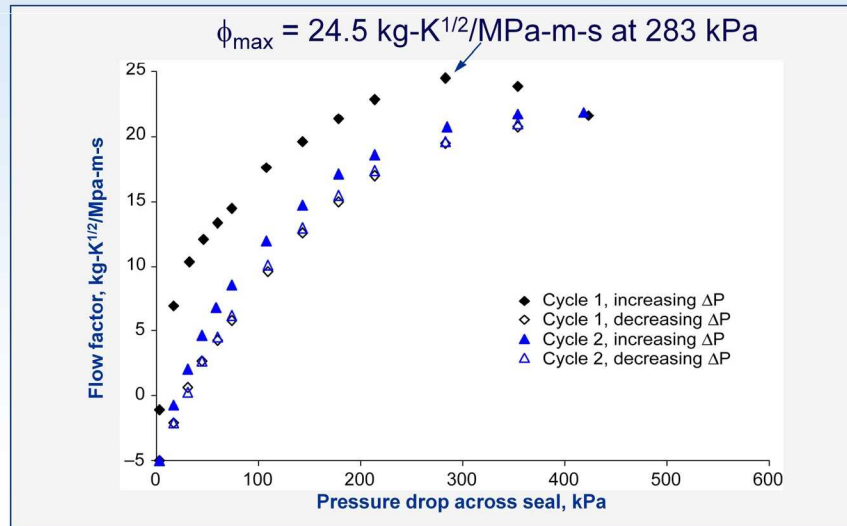Initial Static Leakage Performance at 533 K
Radial Clearance = 48.3 µm

$\phi_{max} = 22.7$ kg-K$^{1/2}$/MPa-m-s at 283 kPa
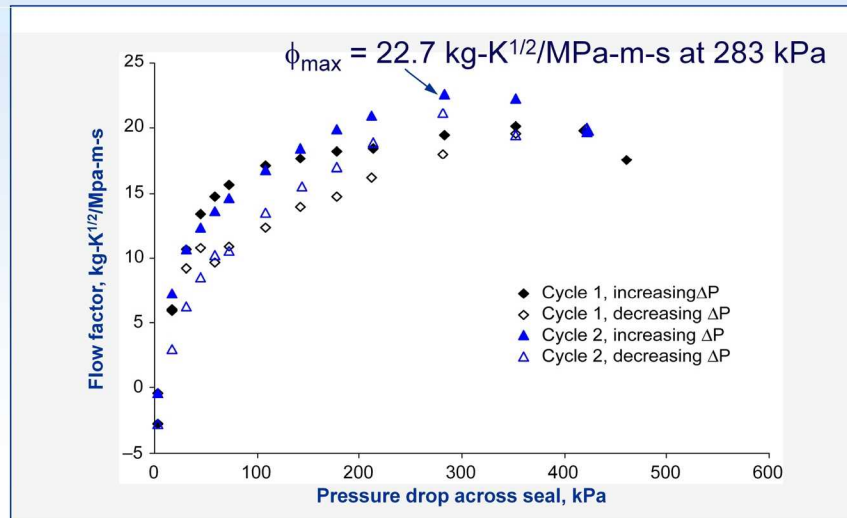
The seal exhibits more hysteresis than at room temperature.

The maximum flow factor occurred at 283 kPa and was 22.7 kg-K$^{1/2}$/MPa-m-s.

This higher flow factor can be attributed to an increase in radial clearance due to the difference in the coefficients of thermal expansion for the seal and rotor materials. At room temperature the radial clearance is 25.4 µm. Assuming that both the seal and rotor are at 533 K the radial clearance increases to 48.3 µm, nearly double the build clearance.

Initial Static Leakage Performance at 700 K
Radial Clearance = 61 μm

ϕ_{max} = 24.5 kg-K$^{1/2}$/MPa-m-s at 283 kPa

This is the initial static leakage performance at 700 K. At this temperature the radial clearance grows to 61 μm. The maximum flow factor occurred at 283 kPa and is 24.5 kg-K$^{1/2}$/MPa-m-s.

Bind-Up Test Part 1–Static Leakage Performance at 320 to 344 K

ϕ_{max} = 13.8 kg-K$^{1/2}$/MPa-m-s at 421 kPa
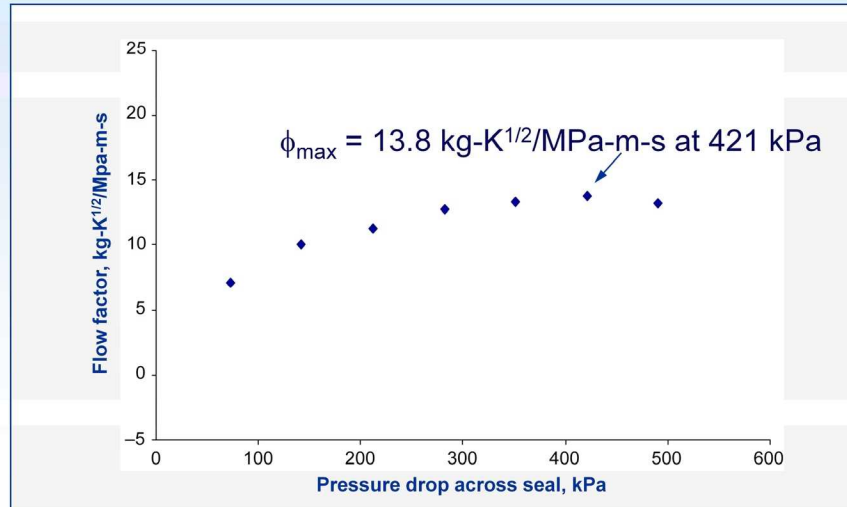
This is the static leakage performance in the Bind-Up test part 1.

The inlet air temperature steadily increased from 320 to 344 K due to residual heat in the insulated piping between the air heater and test rig.
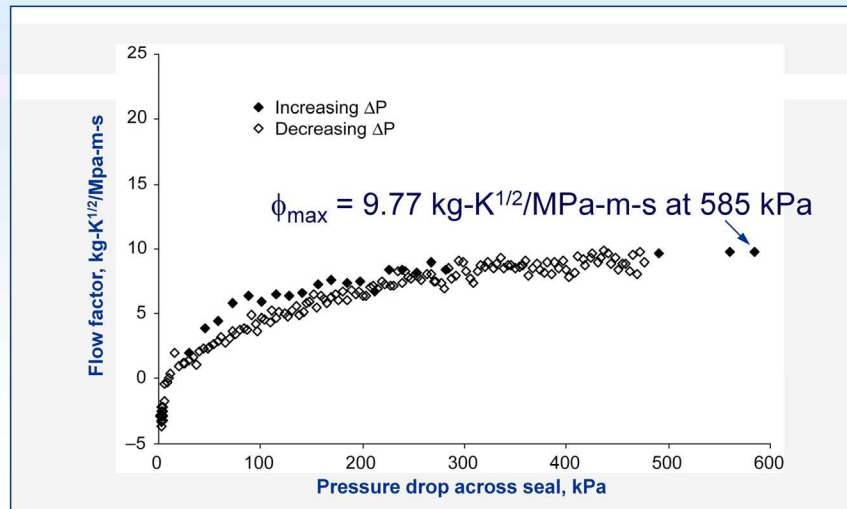
The maximum flow factor of 13.8 kg-K$^{1/2}$/MPa-m-s at 421 kPa is less than the initial static test.

Recall that the shaft is turned by hand at 0 kPa between each data point.

Rotation assists in moving the seal into its optimum position.

Bind-Up Test Part 2–Static Leakage Performance at 342 to 345 kPa

In part 2 of the Bind-Up test the inlet air temperature was 342 to 345 K.

The maximum flow factor was 9.77 kg-K$^{1/2}$/MPa-m-s at 585 kPa and is lower than the flow factor in part 1 of the bind-up test. Recall that in part 2 of the bind-up test that the shaft is rotated by hand at every pressure differential test point up to 276 kPa. This result further demonstrates the importance of shaft rotation to obtaining the optimum seal position.

Repeat Static Test Leakage Performance at 300 K

Maximum Flow Factor is:
- 10.3 to 13.2 kg-$K^{1/2}$/MPa-m-s
- 24 to 41% less than initial test
- 60 to 80% of flow factor for straight 4-tooth labyrinth seal with 229 μm (0.009 in) radial clearance

Legend:
- ◆ Cycle 1, increasing ΔP
- ◇ Cycle 1, decreasing ΔP
- ▲ Cycle 2, increasing ΔP
- △ Cycle 2, decreasing ΔP
- ■ Cycle 3, increasing ΔP
- □ Cycle 3, decreasing ΔP

Y-axis: Flow factor, kg-$K^{1/2}$/Mpa-m-s

X-axis: Pressure drop across seal, kPa
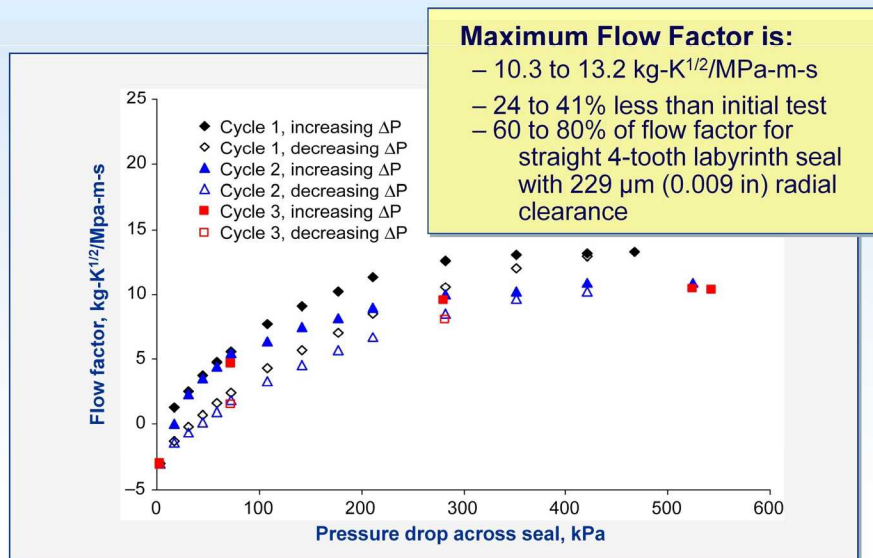
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Self-explanatory.

Unlike the initial static test at 300 K, there is quite a bit of hysteresis.

In all cycles the flow factor is lower for decreasing pressure differential than for increasing differential.

Since there are no temperature changes, the hysteresis is most likely due to internal friction forces within the seal. As pressure differential increases, the fingers toward the rotor due to the pressure blow down effect. It is surmised that friction forces hold the fingers at the smaller clearance as pressure differential decreases resulting in a lower flow factor. Reducing the pressure differential to zero releases the fingers.

The repeat static tests at 533 and 700K had similar hysteresis and 10 to 20 percent lower maximum flow factors than the initial static tests.

Second Spin Test Leakage Performance at 300 K
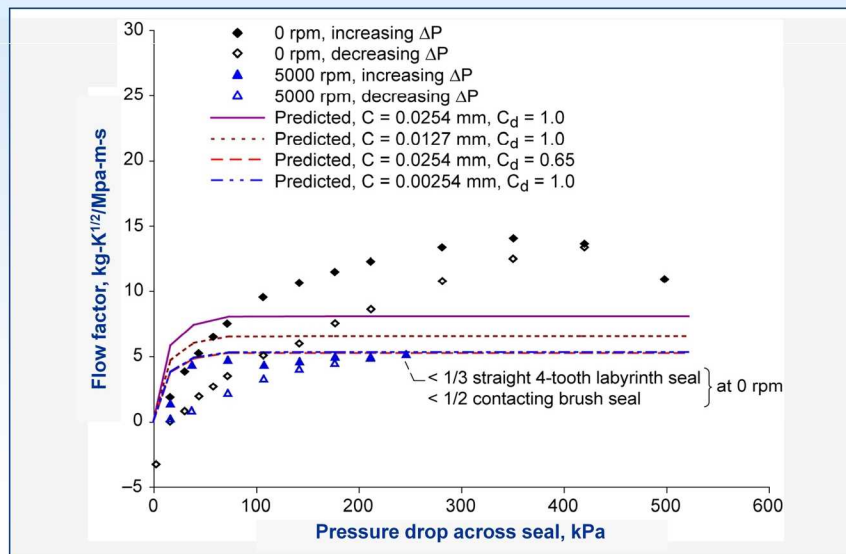
This is the leakage performance of the non-contacting finger seal just prior to and during the second spin test at 5000 rpm.

Predicted flow factors are in reasonable agreement with the data.

Hysteresis is present in both the static and second spin test data.

The flow factor with shaft rotation at 5000 rpm is substantially less than the static flow factor; approximately half at 241 kPa where it begins to level out.

The measured flow factor at 241 kPa was 5.2 kg-$K^{1/2}$/MPa-m-s. This is less than on third of the measured flow factor of a straight four-tooth labyrinth seal and les than on half the flow factor of a contacting brush seal at static conditions previously reported.

The measured flow factor for this non-contacting finger seal is similar to that measured for a contacting finger seal at 186 m/s 700 K, and 276 kPa of 3 to 6 kg-$K^{1/2}$/MPa-m-s.

Second Spin Test Seal Power Loss at 300 K
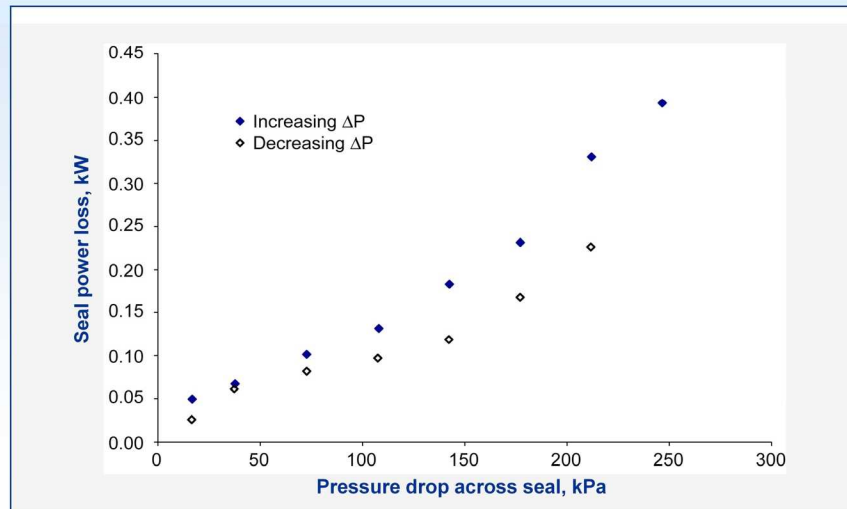
Power loss at 5000 rpm and 300 K increases as a function of pressure drop across the seal.

The maximum power loss at 5000 rpm is approximately 0.4 kW at 247 kPa.

The seal exhibits some hysteresis. Seal power loss for decreasing pressure differentials is approximately 30 percent less than for increasing pressure differentials.

The hysteresis in power loss corresponds to the hysteresis in flow factor data.

This makes sense since lower flow factors indicate smaller clearances. Power loss decreases as radial clearance decreases.

Although a direct comparison can not yet be made, it is observed that the non-contacting seal power loss is of the same order of magnitude as that for the contacting brush and finger seals.

Wear Results After Initial Spin Tests

Seal

- Visual inspection finds seal in good condition.

- No significant change in weight.

- Light burnishing on:
  - All low-pressure lift pads at I.D. near high-pressure edges.
  - High-pressure fingers around the finger "toe".

- All the fingers and lift pads are free to move.

Rotor

- Shiny wear track of uniform axial length around entire circumference has no perceptible depth by touch.

- Grooves were clean and free of debris.

- Burnishing is result of brief contact during start and stop of shaft rotation.

- There was no rapid or substantial rise in seal exit or back face temperature.

Non-contacting operation was achieved.

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Self explanatory.

Conclusions

The Non-Contacting Finger Seal promises low leakage and long life capability.

1. No measurable wear after 93 minutes of rotation at 300 K and 5000 rpm.
2. Non-contacting operation was achieved at 5000 rpm and 14 to 241 kPa.
3. The measured flow factor at 5000 rpm and 241 kPa was
   - <1/3 of the measured flow factor of a straight 4-tooth labyrinth seal and
   - <1/2 of the measured flow factor of a contacting brush seal at static conditions.
4. Rotation is required to properly seat the seal and results in lower flow factors.
5. Non-contacting finger seal power loss is the same order of magnitude as brush and finger seals.

The simplified flow model is in reasonable agreement with data once flow chokes.

Further testing and analysis is needed to
- understand the nuances of this particular non-contacting finger seal design
- develop useful design methodologies and predictive tools.

Fluid-structural modeling is needed to
- understand bind-up observed at 276 kPa
- determine design modifications to achieve higher pressure capability.

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Self explanatory.